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Problems in the Construction of Woodworking Machines

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Problems in the Construction of Woodworking Machines

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ABSTRACT: For the construction of modern wood working machines, problems such as the most favorable bearing of tool spindles and shafts play an important part. The author considers the constructive and functional details of these machines. He discusses the interrelationships of speed, bearing load, fit and lubrication, as well as bearing devices and their installation. He then deals with particulars of bearings of motor shafts which serve at the same time as tool spindles, and also with questions concerning oscillating shafts.

Another, equally important construction detail is the chip suction system. Defects that might occur in the construction of suction hoods or at the machines themselves are described.

Finally, questions of noise abatement, its cause and elimination are discussed.

INTRODUCTION /237*

The design of modern woodworking machines is directed, among other things, to improvement of the tool spindle bearings and to the elimination of disturbing secondary phenomena which cannot be eliminated later, or only with difficulty. The causes are usually the poor efficiency of suction hoods and the noise level radiated away from the machines.

TOOL SPINDLE AND MAIN SHAFT BEARINGS

In general, we consider normal, standardized roller bearings for the tool spindle and main bearings. In the choice of a bearing for a given working condition, the

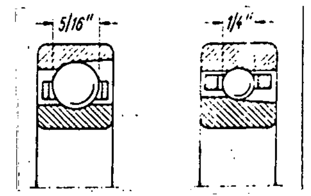
*Numbers in the margin indicate pagination in the original foreign text.

designer must often approach the upper limits of rate of rotation and load. Limiting cases for grease lubrication often exist. These bearings require special checks during operation. as well as maintenance and measures for bearing exchange, in order to avoid failures. In the following, we report working experience with roller bearings in a frequently problematical range of applications. Through discussions of proved bearings, the manufacturers should be stimulated to consider their bearing designs critically. Users should obtain some suggestions for maintenance and bearing exchange, which go beyond what is generally known.

Rate of Rotation

The high cutting rate in wood milling, with upper limits around 100 m/sec, requires correspondingly high rotational rates for the tool spindle. The idea of "high rotational rate" for a rolling bearing must always be understood relatively, because the nominal rotational rate of a 4-pole three phase squirrel cage motor, 1,450 RPM, can already be the allowable rotational rate for a large self-aligning roller bearing, and the general ground rule of design, working rotational rate $\approx$ 2/3 allowable rotational rate, cannot be maintained. If the calculated bearing strengths, or the limiting values from tables, must be exceeded, then rolling bearings with smaller deviations in mass, shape, and position than the usual design must be used. Rolling bearings for very high rates of rotation, with a limiting rate of rotation about 3 times higher than for the normal design, require special types of construction (Figure 1). These bearings have smaller ball diameters, a larger number of balls, and different ball guides, beveling, contact angle, and design of the inner rolling path than the normal bearing design. For radial tapered ball bearings, bearing series 72 . . . Sp. 73 . . . Sp., which are preferably used for bearings on grinder spindles, the allowable rate of rotation can be given by the manufacturer

Figure 1. Tapered ball bearing.
Left: general type of design.
Right: Special design for maximum rotational rates.



only after establishment of the intended design and the working conditions. High-speed roller bearings require high accuracy in the surrounding components, as the state of mounting is decisive for operating safety. Rotational rates above 17,000 RPM, which occur primarily on surface planers, are rare. At higher rates of rotation we must expect higher operating temperatures. Therefore, it is necessary to use bearings with greater radial tolerance. Ball bearings of this type cannot easily be axially prestressed. /238

Load

High loads for the roller bearings arise from high mass of the tool (e. g., the cutter shaft of a cutting spindle mill, which weighs 1,500 kg); or of the work piece (e. g., the 4,000 kg mass of a veneer block in a furniture veneer planer); from mass forces (e. g., 16,000 kp at the dead center position of the crank drive on a saw mill); from cutting forces (e. g., 3,000 kp on a furniture veneer planer with a 2,600 mm working width) and from unbalances in all fast-rotating planer and circular saw spindles, due to operating errors. The selection of a bearing and its fitting to the bearing site on the shaft and in the housing depends on the load actually occurring.

Fitting

The choice of fittings is difficult, as it is not in all cases possible to specify the type of loading unambiguously as a point load or a peripheral load for a particular roller path ring. Heat produced in milling and by high rates of rotation, as well as motor heat from tools fastened to the motor shaft, require axial displacement in any case for a ball bearing used as a floating bearing. In most cases, there is a peripheral load on the inner ring. This type of load requires solid mounting on the shaft and flexible mounting in the housing. During operation there is often additional loading from a rotating force, such as centrifugal force, so that solid mounting is also necessary for the outer ring. The axial shift should then be placed in the housing. To the extent that it is not possible in the design, then a housing fit of K 6 or J 6 must be established for the outer ring of a floating bearing.

Bearing Clearance

The correct clearance is decisive for the lifetime of a roller bearing. High rotational rate and load, and, usually, a high temperature drop between the inner and outer rings – the inner ring is heated from the tool or the motor armature, while the outer ring is cooled by the cool air around the housing – require in every case calculation of the decrease of radial clearance on installation of the bearing. Too little clearance causes whistling bearing noise and allows the bearing temperature to rise very rapidly.

For oscillating needle bearings, 50 μm clearance is necessary. Self-aligning ball bearings for mill shafts, with a bore of 130 to 150 mm, require 130 . . . 150 μm working tolerance.

Lubrication

Most roller bearings are lubricated with grease because the seal design is simpler than for oil. In designing the lubricating system, it is important that the grease to be forced in is unconditionally forced into the bearing, and the used grease displaced. There should be enough space to collect the grease leaving the bearing. The grease should be prevented from rotating. For this purpose, radial ribs or blind holes in the axial direction are placed inside the bearing cover. They hold the grease better than smooth internal spaces. The formation of grease chambers by peripheral rings is undesirable, therefore, even if shaft insertion and removal is made easier by peripheral rings on the floating bearing side.

Rates of rotation which are relatively higher for the bearing size almost always require supplementary lubrication. But the addition of excessive amounts of grease easily causes overlubrication with the results of hot running, decomposition of the grease, dry running, and destruction of the bearing. For most bearings, it is necessary that excess grease be eliminated automatically. Grease amount controllers which centrifuge the displaced old grease out into appropriate containers have proved themselves (Figure 2).

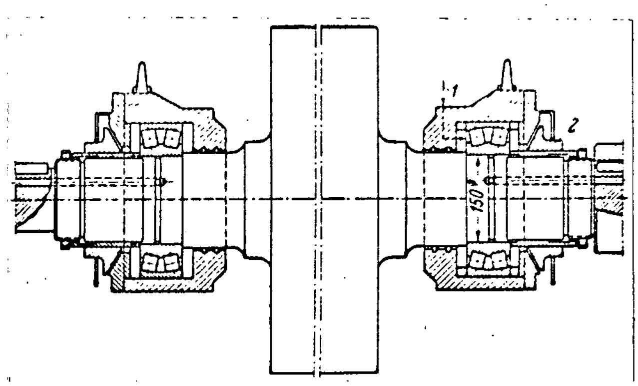


Figure 2. Mill cutting spindle bearings with grease amount controllers. 1: path of the grease forced in. 2. centrifuging disk.

Because of high dust production, the grease must be sealed. The calculated greasing periods must then be decreased by about half. For many bearings on wood-working machines, the greasing is a limiting factor, so that it deserves special care.

Oil lubrication is used primarily when the bearings run along with machine parts for which oil lubrication is necessary, such as extension drives for tool spindles. But the design and maintenance of oil-lubricated bearings require greater expense than for grease lubrication. Oil change and oil circulation lubrication is coming into application. With a fixed installation position, and for shafts arranged vertically and horizontally, labyrinth sealing with oil spinner disks can be considered as shaft seals. Shaft seals are often the cause of severe bearing heating. For satisfactory operation, it is important to have hardened and polished sealing surfaces, good lubrication of the seal lip, and an appropriate sealing lip material such as Perbunan.

Structure

Before bearing selection, one must first determine whether one must expect any deformation of the shaft and bearing housing carrying the roller bearing, or of the mounting surfaces for the bearing housing. Oblique positions of the shaft and housing at the bearing sites, due to working errors or thermal expansion, cannot be borne without damage by non-rotatable roller bearings. In some cases, one bearing housing can be shifted with respect to another for adjustment of the tool spindle position in the radial direction. The oblique position of the shaft produced in this way causes additional forces and rapid bearing failure with rigid bearings. To mention a particularly typical example: one manufacturer uses rigid needle bearings for a grinder cylinder bearing.

Another uses a ball bearing for which the outer ring is arranged in the bearing housing so that it can deflect. Working conditions are identical for both machines. While /239
the deflectable ball bearing exceeds the nominal lifetime, the rigid needle bearing attains only a fraction of it. The lifetimes attained in practice for the rigid needle bearing and the deflectable ball bearing are in the ratio of 1:10 to 1:20.

For a radial bearing which must also accept axial forces in both directions, the outer ring must be established in the axial direction in the housing. The rest of the bearing must be able to shift axially in the bearing or in the housing. Here it is necessary to consider mounting tolerances, shaft bending, and thermal expansion. The extent of shaft heating is often under-estimated, so that the necessary length compensation is not present.

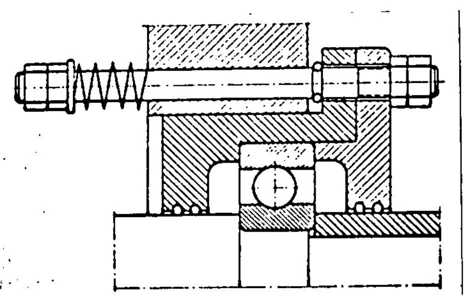


Figure 3. Floating bearing in sliding sleeve.

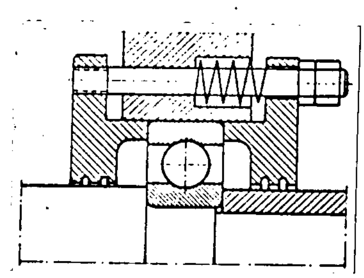


Figure 4. Floating bearing with outer ring clamped at the end side.

To the extent that a floating bearing is not installed in a bearing sleeve which is axially movable and secured against rotation, the outer ring should be solidly clamped with a bearing cover which can slide (Figure 4).

Hydraulic Mounting

Inner rings of roller bearings are fastened to the shafts exclusively with interference fits. During fitting by heating the inner ring, only low mounting forces are required. But rather large forces are applied on removal, even for average bearing dimensions. Repeated removal of the pressed joint can quickly decrease the quality of the shaft seat, so that satisfactory seating of the inner ring is no longer ensured. With bearings exchanged repeatedly, repeated rotation of the inner ring on the shaft, complete bearing destruction, and seizing of the inner ring are observed. For important bearings, as for expensive cutting shafts, hydraulic mountings should be used above about 80 mm bore. The method consists of releasing the press fit of the inner ring by the application of high pressures, producing an oil film in the joint space. This largely prevents contact of the jointed parts, greatly diminishing the separating force and protecting the shaft surface from damage. Mounting time and expense are diminished by hydraulic mounting.

Precision Tool Spindles

Precision tool spindles were developed for use in many processing unit types. The machine units consist of cross- and deflection-support with the precision tool spindle and drive fastened to that (Figure 5). The tool axis can be arranged in any position with a 360° circle, so that the technologically most favorable work procedure can always be established. The development of the precision tool spindle was stimulated by improvement in hard alloy tools and increased requirements for quality of work surfaces.

Along with sufficient rigidity, a tool spindle should have high rotating accuracy, low-play mounting, and long, accurate lifetime, so that manufacturing errors will have the minimum effect on the quality of the work surfaces.

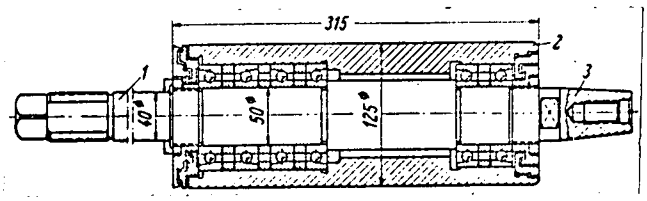


Figure 5. Precision tool spindle. 1. Tool mounting. 2. Housing. 3. Cone for drive pulley.

These requirements are attained through careful selection of materials for the parts, by high manufacturing precision, and by many selected radial tapered ball bearings (Bearing series 72 . . Sp. 73 . . Sp precision ball bearings). In order that bearing tolerance can be adjusted and the wear of the balls and ball races can be compensated for, there is a micrometer screw adjustment by which the outer ball bearing ring can be shifted in relation to the inner ring. This type of design has the advantage that a transition fit can be chosen for the outer bearing ring on the fixed side of the bearing, avoiding rotation of the outer ring in the housing from an additional rotating load due to centrifugal force, as from torn-off cutting plates. If springs are provided for compensating bearing tolerance, then a clearance fit is necessary for the outer ring. It can move peripherally in normal operation. In every case, the number of roller bearings is greater than 2 in order to obtain low loads per bearing and long lifetime. Satisfactory running of 4 to 8 precision ball bearings can be attained only through careful manufacture. Only selected bearings with the same fit and tolerance can be used. The bearing seats must show high precision in shape and position.

The precision tool spindle is clamped at the tool support for faster exchange (Figure 6). In order to avoid errors in maintenance work, there should be fixed intervals at which the tool spindle should go to the manufacturer for cleaning, lubrication, and tolerance checking.

One unit with a precision tool spindle has a greater size than the design with the tool-supporting motor, and is also more expensive. But in return, the otherwise necessary high-frequency drive is not required for rotational speeds above 2,800 RPM, and the bearings are of higher quality.

Machining units are used in multi-side planers and double end profilers. They are produced in a multitude of sizes and designs. Standardization of the major and connector dimensions would bring economic advantages to manufacturers and users, and should be strived for (Figure 7). /240

Tool-Supporting Motors

With these motors, the tools are fastened to the rotor shaft. They have a long, low design, so that circular saws can reach a good depth of cut, and so that planers with small diameters can be used. The range of application is the same as for the precision tool spindles. For surface planers, the motor is driven at high frequency, 200 - 300 Hz. Profiling planers normally have spindle speeds of 5,600 RPM and occasionally 8,000 RPM, and circular saws 2,800 RPM (Figure 8). In the cold state, the bearings of high-frequency motors must have a marked play in the axial direction, and somewhat less in the radial direction. During operation, the shaft is heated from the rotor and the tolerance decreased. Inadequate tolerance in the axial direction, in the cold state, leads to hot running after

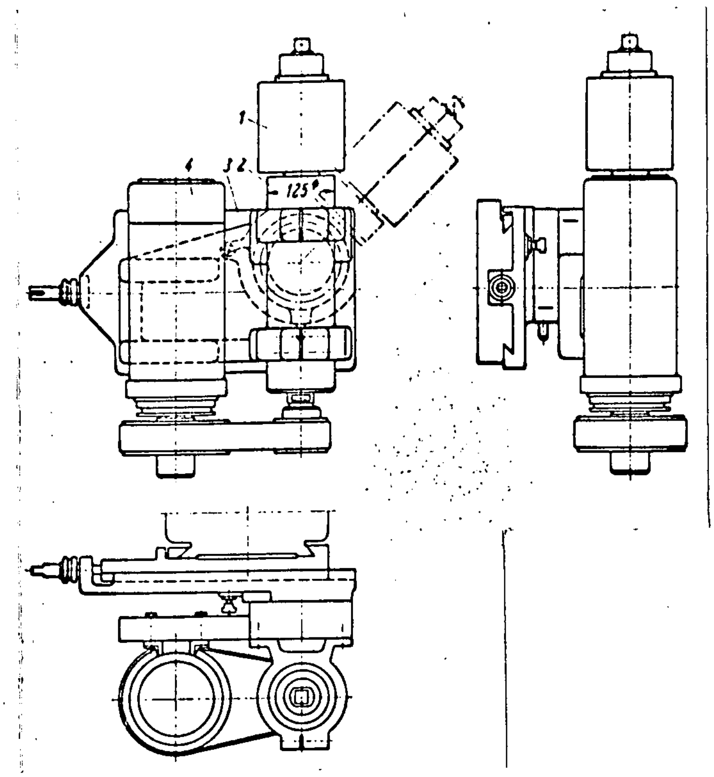


Figure 6. Machine unit with precision tool spindle. 1. Tool. 2. Precision tool spindle. 3. Cross and rotary support. 4. Drive motor.

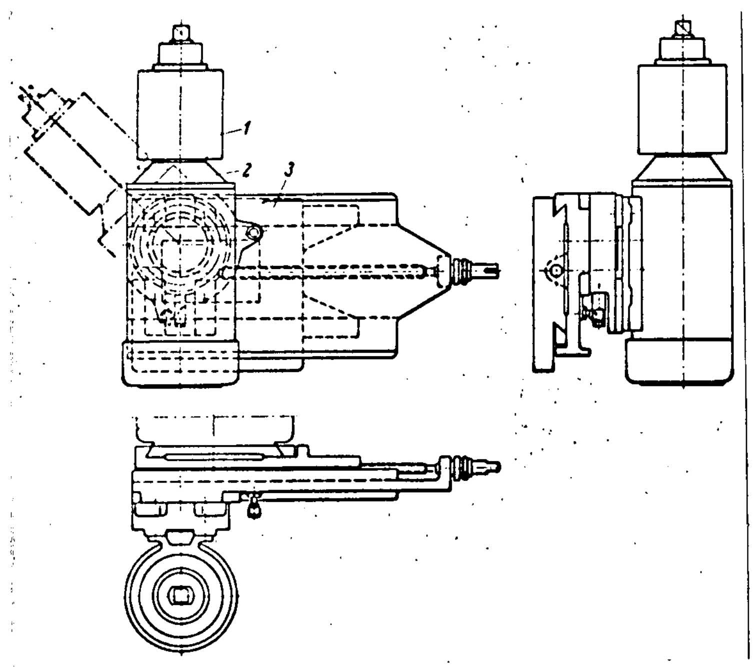


Figure 7. Machine unit with tool-support motor. 1. Tool. 2. Tool-support motor. 3. Cross and rotary support.

a short time for surface planer motors, which are loaded primarily axially.

Motors for planing tools and circular saw blades usually have a built-in disk brake, so that they can also be used to drive precision tool spindles, but with simpler bearing design.

Oscillating shafts

Oscillating shafts, grinder cylinders, and grinder spindles are special bearing cases which could not for a long time be solved satisfactorily. In roller bearings of the normal type, the support and axial displacement of the entire bearing housing in sliding bearings require high construction cost. Cylindrical roller bearings and needle bearings without guide cages tend toward binding and blocking in the axial direction. With an appropriately large working tolerance in full-needle needle bearings, it is possible to keep the binding phenomenon which occurs from blocking, because the bearing needles straighten out again in the unloaded zone, but the running accuracy is not enough for a grinder cylinder. The binding forces which occur cause increased operating temperatures and more severe air noise. Needle bearings with a guide cage prove to be better. The axially parallel guiding of the needles avoids binding and allows the use of bearings with limited working play. Rigid needle bearings are very sensitive to axial misalignment. They should be installed in a clampable radial bearing in which the clamping must be loosened for every adjustment of the grinder cylinder.

A special ball bearing from the Swedish ball bearing factory, SKF, has proved to be distinguished (Figure 9). By use of many balls, the proportionate load and flattening of the balls and races is diminished. But this bearing is sensitive to shaking in the stopped condition.

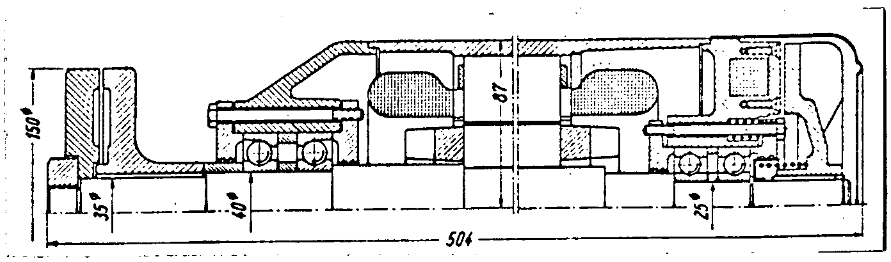


Figure 8. Tool-supporting motor.

All roller bearings are subjected to severe abrasion if the shaft is moved back and forth axially without rotation. The lubrication periods stated in handbooks and catalogs cannot be applied for oscillating bearings. The oscillation forces the grease out of the bearing, so that filling the grease supply is necessary for most bearings after only 48 running hours.

SUCTION

In part, suction removal of chips in woodworking machines works at extremely low efficiency, because there have been few scientific observations. Improvement of catching chips and dust in the machine, as well as of /241 the efficiency of collecting hoods and fresh air channels in machines are very important problems.

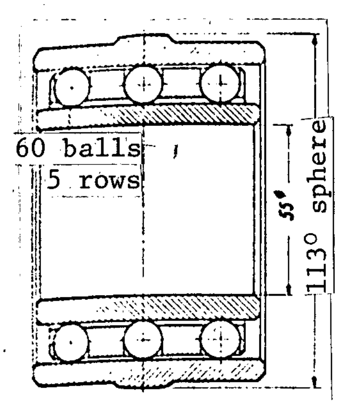


Figure 9. 5-row ball bearing for oscillating shafts.

Collecting Hoods

Suction and collecting hoods are met in quite different forms. Their action is often very inadequate. In order to attain extensive collection of the chips, it is necessary to know the flight direction of the chips. It is generally assumed that the chips fly away in the direction of the tangent to the cutting circle where the cutters emerge from the wood. This assumption applies only for circular saws with a negative rake angle, which are scarcely ever encountered any more, and for grinding cylinders. For tools with a positive rake angle, the chip will be moved peripherally by the cutting adge after it is separated, until it separates from the tool because of centrifugal force. The scattering angles shown in Figure 10 appear as a result. The scattering angle is larger for planer chips than for circular saw chips and grinder dust. The cross-hatched angle shows the scattering of the main bundle. With this knowledge, the designer has the possibility of choosing the shape of the collecting hood so that the chips are collected and picked up by using the speed that they received from the tool. But the construction of a machine does not in every case allow the collecting hoods to be designed so that the average direction of the chip flow coincides with the center of the suction column, so that all the chips go directly to the suction pipe. In this case, the collecting hood must be designed so that there is a high air velocity in the zone where the chips strike, to pull part of the chips into the suction pipe. In a mixture of chips, the particles in the dust fraction are rather light and can reliably be pulled away. But drawing off coarse chips requires air flow velocities which are not practically usable in conical collecting hoods, so that the velocity losses from chip impact must not be too great. The impact angle should not exceed 30°. Otherwise it is hardly possible to use the kinetic energy of the chips for their transport.

Often the chip collecting hood serves simultaneously as a shield to prevent touching of the tool. Then, for reasons of working safety, the designer tries for the most complete coverage possible, with less optimal suction conditions. Here, it is often forgotten that fresh air must flow into the collecting hood. Often considerable excessive air velocities occur in parts of the collecting hood where they cannot be effective. The in-flow conditions are often so bad that turbulence is produced because the direction of the in-flowing air is opposed to the direction of tool rotation over a long distance. Chips can even be thrown out of the air intake opening due to

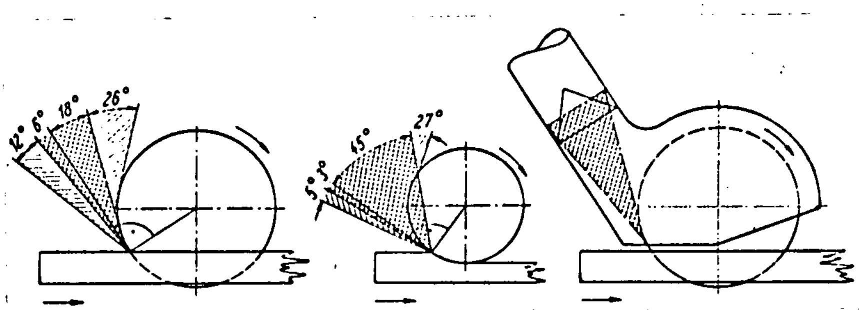


Figure 10. Scattering angles for chips. Left: circular saw. Center: planer. Right: suction column in the main chip stream for a circular saw.

turbulence development. For example, such a distinctly unfavorable situation occurs with the collecting hood of Figure 11. The chips strike the wall perpendicular to the work surface at a poor angle, so that almost all their kinetic energy is lost. There is only a slight air flow at the point of impact, so that a large part of the chips which drop out of the stream are carried out of the collecting hood by the work piece.

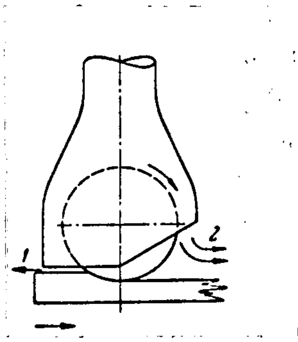


Figure 11

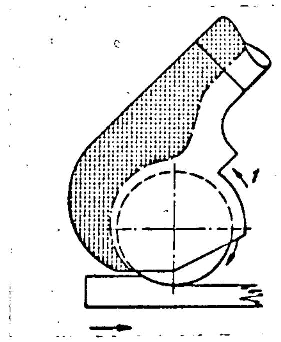


Figure 12

Figure 11. Poor collecting hood shape on a planer. Chips come out at 1 and 2.

Figure 12. Good collecting hood shape on a planer. 1. Supplementary air inlet. The shaded field shows the course of the main chip flow.

By means of a spiral-shaped contour for the impact wall of the collecting hood, a directed chip flow can largely be achieved (Figure 12). Most of the in-flowing air is not affected by the tool. This collecting hood shape is practical for planers, because here the suction column must be displaced from the average direction of chip flow for reasons of space.

For grinders, by use of a small collecting slot, it is possible to attain a sufficiently high collecting velocity in the collector, even with large width of the /242 work, so that the grinding dust will certainly be drawn away (Figure 13). Impractical collecting hood shapes are often found on machines with large working width, through which a large supply stream enters rather ineffectively.

The suction action for belt grinders is often quite inadequate, although there are many recommendations for collecting the grinding dust at the work site. In most cases, there is only a collecting hood over the driving belt roll, and its action is decreased by the turbulence produced by the belt roll. This disadvantage can be eliminated by pipes in the side walls of the collecting hood, into which the belt rolls enter, so that the bottom and inside of the rim are covered. (Figure 14.)

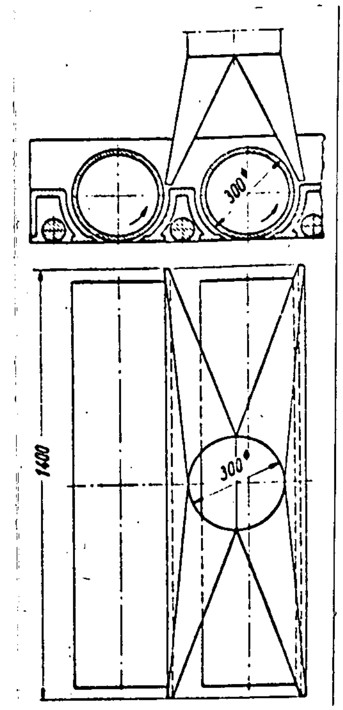


Figure 13. Collecting hood for large work widths, carried out to 2,600 mm.

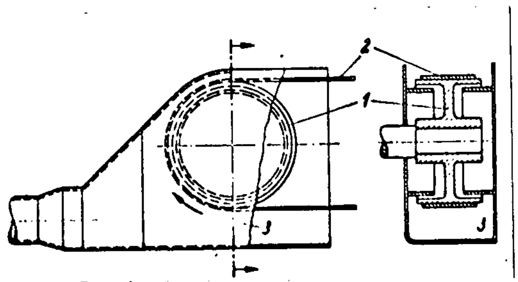


Figure 14. Collecting hood for a normal belt grinder.
1. Belt roll. 2. Grinding belt. 3. Collecting hood.

Inadequacy of the Suction Machines

In many cases, the centrifugal blower is properly dimensioned for the carrying flow, but in operation it has too little pressure and as a result, a greatly decreased air capacity. The cause is the design of the machine, which allows air flow only to a limited extent. There results a very high resistance, so that the centrifugal blower works practically against a closed damper and the dynamic proportion of overpressure, Δp_{dyn} becomes ≈ 0.

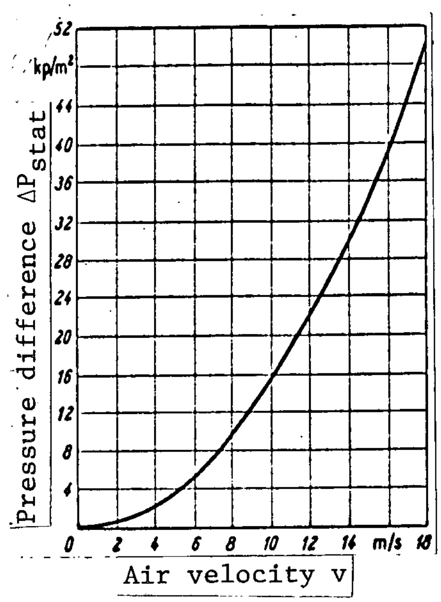


Figure 15. Resistance parabola for the collection hood of a disk sander.

On machines with fresh air suction channels, there are often unfavorable in-flow conditions due to impractical shape, from the viewpoint of flow technology, or inadequate cross section. The practical effects are turbulence development, high resistance, and considerable overvelocity. It would be a pressing problem to improve the intake flow conditions at the machines to ensure the necessary carrier flow into the pipes and not to come to the range of static pressure loss, which the use of high-pressure blowers requires.

So that the designer can establish an aerodynamic system with optimal suction volume, considering all the existing influences, the resistance characteristics of the machines must be known. Figure 15 shows the resistance at a disk sander with 800 mm disk diameter, determined by throttling the air speed after a 3,000 mm long smooth tube connected to the grinding dust collection hood. In general, the measurement cannot be made immediately after the collecting hood because there must be a sufficient stabilizing section for the flow. The smooth pipeline should have a length about ten times the pipe diameter. For measurement of air velocity over 3 m/sec in pipe lines, one generally uses the Prandtl impact tube.

Without knowledge of the static pressure loss at the suction machine, the possibility of checking and calculating new aerodynamic systems is lacking, and there is danger of under- or over-dimensioning.

PROBLEMS OF NOISE ABATEMENT

We have previously reported on the basic technical possibilities of noise abatement in machine woodworking (Schmutzler, 1967). The following section treats special problems of sound transmitted through solids and air, which determine the sound level for some machines.

High speed and power require great rigidity of the entire machine system. Individual welded machine stands are used, in order to utilize the advantage of the elasticity modulus of steel, which is 3 times as high as for gray cast iron. But, through the transition from cast to welded construction, the machine becomes more liable to vibration, and it is not rare for resonances from the alternating forces of milling to jar the machine table also. With welded designs, experience from light construction must be used in order to attain high rigidity. Experience is often lacking to give the machine the dynamic rigidity necessary for smooth running. Also, the machine stands are often built so that the mechanical resonances cannot be calculated in advance.

Solid sound transmission usually cannot be avoided by increasing the mass alone. The mass increase must be in the proper position. In particular, it is important to corrugate large-area stand walls with little static loads, in order to exclude resonance. Of course, a machine which is too loud can be installed in a noise-blocking shell; but this is not cheap, because of the high investment and manual labor.

Body Sound Vibrations of Machine Stands and Tables

The alternating forces occurring in the milling process often excite the parts rigidly bonded to the machine stand and table, such as the tool spindle bearings, work-piece supports, or the counter-knife in planing machines, especially with light welded designs, so that sound vibrations are transmitted through the solid and radiated off as air noise. /243

It is difficult to attain damping by antidroning coatings on steel plates 10 to 15 mm thick. One appropriate noise abatement measure is to bolt sound-blocking and sound-absorbing plates onto the welded construction. Individual plates should be made with the largest possible dimensions. Unavoidable joints should be sealed with rubber, and the entire covering is insulated from solid sound transmission on the stand.

The sound-blocking and sound-absorbing plates consist of an outer shell of 1 - 1.5 mm thick steel plate, an absorbing coating of mineral fiber 50 mm thick, and a 1 mm thick perforated plate covering with a ratio of hole area/total surface greater than 30/100. At curves of the housing, the covering must be flexible mineral wool 80 mm thick, compressed by the plates to 50 mm thick. A few supports of thin plate may be used between the shells, only if needed. So that the mineral fibers will not fall out of the perforated plate, a covering of cable nettle cloth

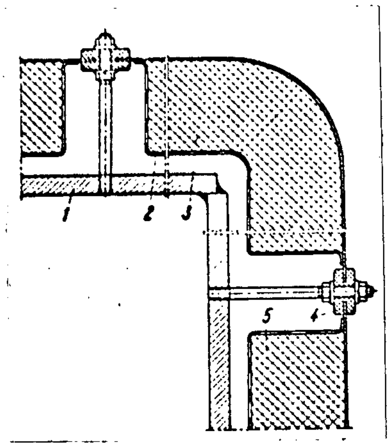


Figure 16. Body-transmitted sound insulation for a machine stand. 1. Machine stand. 2. Perforated plate. 3. Mineral fiber wool. 4. Rubber insulation. 5. Mineral fiber plate.

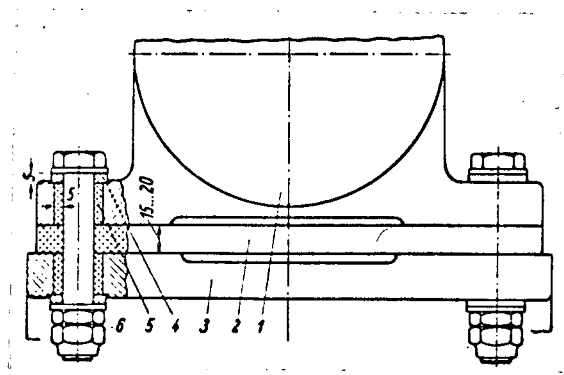


Figure 17. Vibration insulation for a cutter shaft bearing. 1. Bearing housing. 2. Rubber plate. 3. Machine stand. 4. Rubber disk. 5. Rubber casing. 6. Elastic stop nut.

is needed, depending on the fibers used. The distance from the surface of the machine housing to the perforated plate should be 10 to 20 mm. We can consider 10 dB (A) to be a guide value for the decrease of the noise level radiated off the machine housing. For the success of these measures, it is important to keep the maintenance openings closed continuously during operation. Where there are even small holes in the covering, the improvement can be considerably diminished, as the high energy density between the stand and the capsule wall is only partly decreased by absorption, and the remaineer is radiated off through openings in the outer blocking shell, so that the total improvement is not in a good proportion to the expense.

Welded stand designs with large areas of unstiffened walls are almost always excited to transmit sound by solid vibration with machines having cutting shafts. If experience is lacking for stiffening the design with box ribs to prevent vibration, then a sound-damping and sound-absorbing coating must be planned during the design. Otherwise, if it is added later because of a given construction, it must usually be done through expensive compromise solutions. (Figure 16.)

In order to eliminate the transmission of vibrations from cutter shafts, counterknives, and the like to the machine table, the mountings should be vibration-isolated with rubber plates (Figure 17). The rubber must be selected soft enough to ensure acceptable operation of the machine. It must be noted that screws also must be insulated from solid-transmitted sound.

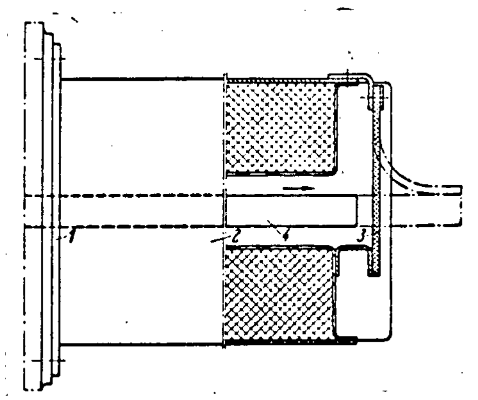


Figure 18. Sound-absorbing lock.
1. Machine stand. 2. Sound-absorbing lock.
3. Rubber plate. 4. Work piece coming out.

Sound-Absorbing Locks

Various machines allow the addition of sound-absorbing locks to the workpiece inlet and outlet sides. The mineral wool layer between the supporting plates must be some 100 mm thick. So far as possible, the remaining openings should be closed off with a rubber apron which is pushed back by the work piece. If necessary, double Plexiglas windows can be used for observation of the work piece inlet (Figure 18).

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